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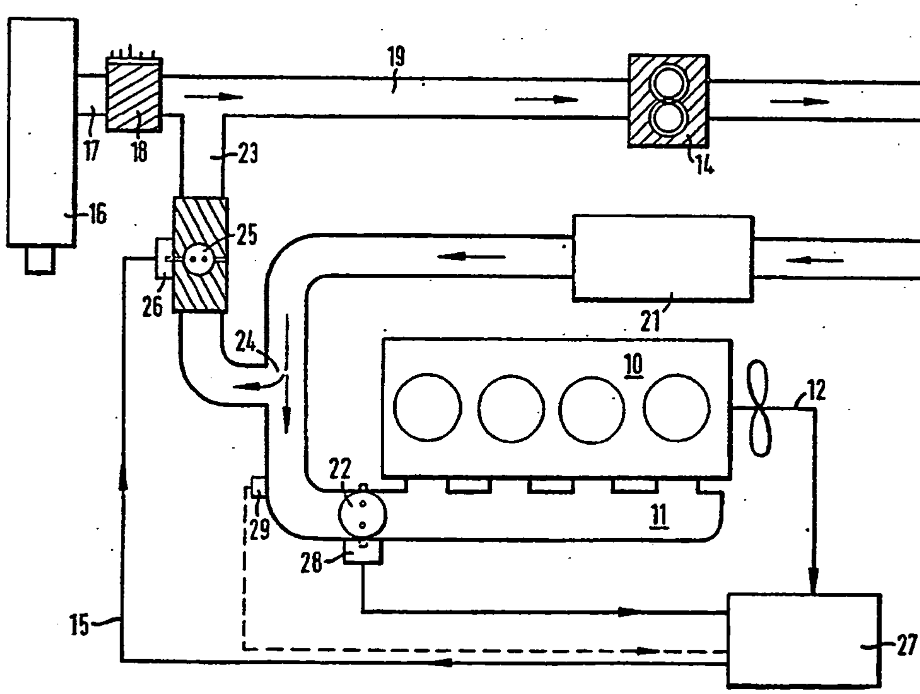
(54) A supercharged I.C. engine air/fuel induction system

(57) The supercharger (14) has a recirculation bypass (23) with a recirculation control valve (25) which is controlled by a microprocessor (27) in response to engine speed and preferably also in response to the setting of the throttle valve (22) and the pressure between the supercharger and the throttle valve to be closed at high engine loads.

The drive ratio of the supercharger may be controlled by the microprocessor.



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The drawing(s) originally filed was (were) informal and the print here reproduced is taken from a later filed formal copy.

combustion air/fuel induction system.

The sketch shows an internal combustion engine 10 having an intake manifold 11, a crankshaft 12, an air/fuel induction system 13 including a supercharger 14, and a boost pressure control circuit 15.

The air/fuel induction system 13 comprises an air cleaner 16 having a clean air outlet 17 connected to an upstream end of an air supply passage of a single point air/fuel metering unit 18 which incorporates fuel injection means.

The downstream end of the air supply passage of the unit 18 is connected to an inlet of the supercharger 14 by one part 19 of an air/fuel induction passage which leads to the intake manifold 11 through the supercharger 14 and an inter-cooler 21 which is downstream of the supercharger 14. The usual driver-operable throttle valve 22 is provided at the junction of the air/fuel induction passage and the intake manifold 11.

A recirculation passage 23 interconnects one location 24 in the air/fuel induction passage between the inter-cooler 21 and the throttle valve 22 with another location in the induction passage part 19. A bypass control valve 25 is provided in the recirculation passage 23 to control flow through the recirculation passage 23 from the location 24 to the passage part 19. Operation of the control valve 25 is controlled by an electrically operable servomotor 26.

The boost pressure control circuit 15 comprises a programmable microprocessor 27 connected to receive input data in the form of an engine speed signal conveniently generated by rotary speed sensing means operatively associated with the crankshaft 12, a throttle position signal conveniently generated by a rotary potentiometer 28 operatively associated with the spindle of the driver-operable throttle valve 22, and a boost pressure signal generated by a pressure sensitive transducer 29 in the air/fuel induction passage between the location 24 and the driver-operable throttle valve 22. The microprocessor 27 is programmed to output an operating signal to control operation of the servomotor 26 and thereby modulate the bypass control valve 25 to a predetermined setting and thus control air flow through the recirculation passage 23 in accordance with the input data received by the microprocessor 27. Thus the pressure of the air/fuel mixture fed to the driver-operable throttle valve 22 is dependent upon the input data fed to the microprocessor 27 and the programme with which the microprocessor 27 is programmed. In practice the programme would be arranged so that the pressure would be high when the engine load demand is high and would be low when the engine load is light.

In another embodiment a supercharger in an air/fuel induction system of an internal combustion engine is provided with variable means operable to vary the drive ratio of the supercharger, the setting of the variable means being determined by an output of a microprocessor which receives input signals indicative of certain operating parameters of the engine, such as throttle angle and engine speed. The microprocessor may be a matrix type electrical control unit.

CLAIMS

1. An i.c. engine air/fuel induction system comprising an induction passage, an operator-operable throttle valve for controlling mass flow through the induction passage to the engine, and supercharging means in the induction passage upstream of the throttle valve operable to boost the pressure of sluid fed to the throttle valve when the required engine power is higher than that of the engine when naturally aspirated, the supercharging means including a recirculation control valve which interconnects the inlet and outlet of the supercharging means and which is operable to control the output of the supercharging means, wherein the recirculation control valve is operable automatically in response to engine speed.

2. An i.c. engine air/fuel induction system according to Claim 1, wherein the recirculation control valve is operable in accordance with the setting of the operator-operable throttle valve as well.

3. An i.c. engine air/fuel induction system according to Claim 2, wherein the control valve is controlled electronicaly by means including a microprocessor and which receives input data derived from an engine speed sensor and a throttle position sensor operatively associated with the operator-operable throttle valve, and which is programmed to output a signal to a servomotor, the servomotor being operable to modulate the recircultion control valve and set it at a predetermined position appropriate for the sensed operator-operable throttle valve position and engine speed.

4. An i.c. engine air/fuel induction system according to Claim 3, wherein the microprocessor is also connected to receive input data from a boost pressure transducer which is operable to sense pressure of flow in the induction passage between said supercharging means and the operator-operable throttle valve and to emit a signal to the microprocessor derived from that pressure.

5. An i.c. engine air/fuel induction system according to Claim 3 or Claim 4, including a feedback loop by which a signal indicative of the position of the recirculation control valve is fed to a respective input of the microprocessor.

6. An i.c. engine air/fuel induction system comprising an induction passage, an operator-operable throttle valve for controlling mass flow through the induction passage to the engine, and supercharging means in the induction passage upstream of the throttle valve operable to boost the pressure of fluid fed to the throttle valve, wherein the supercharging means are provided with variable means operable to vary the supercharger drive ratio and control means operable to control the variable means, the control means including a microprocessor programmed to output a control signal and thereby effect setting of the variable means for a predetermined supercharger drive ratio, in response to input data indicative of certain sensed engine operating perameters such as operator-operable throttle valve angle and engine speed.

7. An i.c. engine air/fuel induction system

compressor adjusted to delivery, the no-load delivery air-charge compressor has a substantially reduced power absorption in relation to its rated operating point. This is due to the low compression ratio arising merely from flow losses and flow separation losses, and from the air mass flow occurring at the instantaneous rotational speed of the rotor. Nevetheless, the power absorption of the no-load delivery air-charge compressor has a level which noticeably impairs the efficiency of the turbocharging assembly.

Therefore, it is the object of the invention to provide for a turbocharging assembly with controllable air-charge compressors, which are in driving connection with a driven means which cannot be cut out during operating periods of the associated internal-combustion engine, a minimisation of the power absorption of the air-charge compressor adjusted to no-load delivery.

The invention is characterised in that upstream of the air inlet each controllable air-charge compressor has a multi-stage controllable distributor device, in that between the distributor device and the air inlet there is provided a device for controlling the air flow acting on the rotor inlet, and in that at least one gas-supply duct is connected to the inlet side of each distributor device.

The advantages achieved with the invention lie in particular in that the flow separation losses occurring per se during no-load delivery of an air-charge compressor as a result of defective inflow to the rotor can be eliminated by influencing the air flow at the rotor inlet, that it is possible to

achieve almost zero power absorption of the no-load delivery air-charge compressor, in that the increase in power output achieved with the gas discharge at the outlet of the exhaust-driven turbine substantially compensates for the existing power absorption of the air-charge compressor with no-load delivery during no-load operation of the internal-combustion engine, in that mechanical decoupling between the air-charge compressor and its driven means can be dispensed with, and in that rapid operating readiness is attained during the transition into the cut-in condition of the air-charge compressor.

In the accompanying drawings:

Figure 1 shows a turbocharging assembly with controllable air-charge compressors, an exhaust-driven turbine and compressed air discharge;

Figure 2 shows the compressor performance graph for a controllable air-charge compressor;

Figure 3 shows a turbocharging assembly with controllable air-charge compressors, an exhaust-driven turbine, exhaust gas supply by way of a gas cooler to the cut-out air-charge compressor and an exhaust-gas extraction compressor; and

Figure 4 shows a turbocharging assembly with controllable air-charge compressors, an exhaust-driven turbine, exhaust gas supply by way of the air-charge intercooler to the cut-out air-charge compressor and an exhaust-gas extraction compressor.

A forced-induction internal-combustion engine

(not shown) is supplied with a pre-compressed air charge through an air-charge manifold 11 by an exhaust-driven, freewheeling turbocharging assembly 12 (see Figure 1). The turbocharging assembly 12 is of single-shaft construction and comprises an exhaust-driven turbine 15, a controllable first air-charge compressor 16 and also a controllable second air-charge compressor 17, both of which are in constant driving connection with the exhaust-driven turbine 15.

The pressure connections 18, 19 of the air-charge compressors 16, 17 are connected to the inlets of a change-over device 20 which is formed by two change-over slide valves 22, 23 connected to a duct 21. The change-over device 20 controls the operating condition (no-load delivery or delivery operation) of the two air-charge compressors 16, 17, and also the operation of the internal-combustion engine with one-stage or two-stage air-charge compression. The pressure connection 18 of the first air-charge compressor 16 leads to the inlet of the first change-over slide valve 22, one outlet of which is connected to the gas-supply duct 24 which leads to the air inlet 14 of the second air-charge compressor 17. The pressure connection 19 of the second air-charge compressor 17 leads to the second inlet of the change-over slide valve 23, which has a venting outlet 26, and to the other outlet of which the air-charge manifold 11 is connected. A gas-supply duct 25 branches from the pressure connection 19 and leads to the air inlet 13 of the first air-charge compressor 16.

Respective three-stage controllable distributor devices 27, 28 are arranged upstream of the air

inlets 13, 14 of each air-charge compressor 16, 17. The distributor devices 27, 28 are jointly connected at the inlet side of the air-intake duct 31. As shown in Figure 1, the distributor device 27 is also connected at the inlet side to the gas-supply duct 25 and the distributor device 28 is connected to the gas-supply duct 24.

In each of the two air-charge compressors 16, 17 and between the respective distributor devices 27, 28 and air inlets 13, 14 a respective device 29, 30 is arranged which enables the direction of the air flow to the rotor inlet to be controlled. Each of the devices 29, 30, which are of like construction, has three separate flow paths "G", "M", "N", each of which can be controlled by an outlet of the distributor device 27, 28. The flow paths "G" and "M" of both devices 29, 30 are provided with adjustable restrictor devices 34, 35, 36, 37 which enable the passage cross-section to be adjusted to the instantaneous air mass flow. The flow paths "G", "M", "N" respectively unite to form the air inlets 13, 14 just upstream of the rotor inlet of the air-charge compressors 16, 17.

Only one of the three flow paths is operative in each case for the operation of the two air-charge compressors 16, 17. When there is admission to the flow path "G", a so-called counter-swirl at the rotor inlet is induced, the flow direction of which is directed counter to the direction of rotation of the rotor. Admission to the flow path "N" induces a non-swirling, that is neutral inflow to the rotor corresponding to the design point of the rotor blades. When there is admission to the flow path "M", so-called co-swirling takes place at the rotor

inlet, the flow direction of which is the same as the direction of rotation of the rotor.

Under no-load and low load the internal-combustion engine has only a small air-charge requirement which is satisfied by the second air-charge compressor 17 alone. Therefore, in this operating phase the entrained first air-charge compressor 16 is adjusted to no-load delivery. No-load delivery is achieved by switching the pressure connection 18, by way of the change-over slide valve 22, the duct 21 and the change-over slide valve 23, to connect it to the venting outlet 26. In this way, although the rotor of the turbocharging assembly 12 is rotating, no appreciable delivery pressure can build up in the pressure connection 18 of the first air-charge compressor 16. At the same time, the passage from the pressure connection 18 to the gas-supply duct 24 in the change-over slide valve 22 is closed; the distributor device 28 for the second air-charge compressor 17 is switched to connect the air-intake duct 31 and the air inlet 14 (switch position "L"), and the pressure connection 19 of the second air-charge compressor 17 in the change-over slide valve 23 is connected to the air-charge manifold 11. Therefore, the second air-charge compressor 17 alone serves for the air-charge supply to the internal-combustion engine.

The compressor performance graph illustrated in Figure 2 shows the principal shift of the compressor operating point of an air-charge compressor 16, 17 adjusted to no-load delivery under the influence of the measures described below. The pressure ratio "p1/p2" is plotted on the y-axis and the air vloume flow "V" is plotted on the x-axis. Like line

patterns denote the respectively associated curves for constant efficiency 51, like rotational speeds 52 of the rotor and the surge point 53. In switch position "V" of the distributor device 27, that is with an open passage from the air-intake duct 31 to the air inlet 13, the air-charge compressor 16 adjusted to no-load delivery would operate at the operating point "A" of the compressor performance graph according to Figure 2.

The power absorption "P" of an air-charge compressor adjusted to no-load delivery is given by the following equation:

$$P = \frac{V * (p1 - p2)}{\eta}$$

In this formulae "$\dot{V}$" designates the air volume flow, "(p1 - p2)" designates the pressure increase and "η" designates the efficiency of the air-charge compressor 16. The formula demonstrates that a reduction in the air volume flow or the pressure increase and/or an improvement in efficiency results in a reduction in power absorption.

The improvement which can be achieved in each case is illustrated by means of the compressor performance graph and for the case in which the first air-charge compressor 16 is adjusted to no-load delivery. The same assertions also apply to the case in which the second air-charge compressor 17 is adjusted to no-load delivery and the first air-charge compressor 16 by itself satisfies the air-charge requirement under low-load of the internal-combustion engine.

If the flow path "M" is operative in the device 29 and, therefore, the air drawn in by the no-load delivery first air-charge compressor 16 enters the rotor which co-swirling, the operating point is shifted from "A" to "B" in the compressor performance graph. At point "B" the compression ratio and the air mass flow are substantially reduced with respect to point "A".

If the passage-cross section for the air mass flow is additionally optimised by the restrictor device 34 situated in the flow path "M", the operating point is shifted further from "B" to "C" towards lower power absorption. The compression ratio at point "C" is in fact increased once more with respect to "B" but the air mass flow has undergone a further substantial reduction. The no-load delivery first air-charge compressor 16 is thereby operating at optimum efficiency at point "C".

Another possibility for decreasing the power absorption of the first air-charge compressor 16 adjusted to no-load delivery is provided if, in addition to all the other measures already described, the air still passing through during no-load delivery does not have to be drawn in but is fed into the air inlet 13 as compressed air. This measure is illustrated in Figure 1. The discharge of compressed air takes place at the pressure connection 19 of the second air-charge compressor 17 by way of the gas-supply duct 25. In position "L" of the distributor device 27 the compressed air then passes along the flow path "M" into the air inlet 13, when the restrictor device 34 is correspondingly adjusted. In that case the shifting of the operating point from "C" to "D" for the air-charge compressor

16 takes place in the compressor performance graph.

A further reduction in the power absorption of the no-load delivery first air-charge compressor 16 is achieved if the intake air of the second air-charge compressor 17 adjusted to delivery operation enters the rotor with counter-swirl along the flow path "G" of the device 30. This measure results in the charge pressure required by the internal-combustion engine being reached at a lower speed of rotation of the turbocharging assembly 12. The result of this is that, in all previously described measures, because of the lower speed of rotation of the rotor, the effective power absorption of the no-load delivery air-charge compressor 16 decreases even further. The compressor operating points respectively designated "A'", "B'", "C'" and "D'" are then given in the compressor performance graph according to Figure 2.

All the measures described above relate to the operating condition of the internal-combustion engine during no-load and low-load. If the internal-combustion engine is operating under partial load, a higher air mass flow is required for the air-charge supply which at this operating point of the internal-combustion engine can be provided by the first air-charge compressor 16 alone, which is designed for higher output; namely, during partial load the second air-charge compressor 17 is adjusted to no-load delivery and the first air-charge compressor 16 is adjusted to delivery.

For this change over, the change-over slide valve 22 changes from position "L" into position "T"; and the change-over slide valve 23 changes into the

other switch position. The two distributor devices 27, 28 also change from positon "L" into position "T".

All the measures described above for reducing the power absorption of the no-load delivery first air-charge compressor 16 are, after this change over, just as effective for the second air-charge compressor 17 now under no-load delivery during partial-load. The compressed air supply to the air inlet 14 of the second air-charge compressor 17 takes place through the gas-supply duct 24 which is connected to an outlet of the change-over slide valve 22 and thus to the air-charge delivery of the first air-charge compressor 16.

Under full-load operation of the internal-combustion engine the two air-charge compressors 16, 17 are connected in series for two-stage air-charge compression. The change-over slide valve 22 and the distributor devices 27, 28 are then shifted into position "V", whereas the change-over slide valve 23 is changed back into its original position as shown in Figure 1.

The compressed-air discharge described above in the air-charge compressor 16 or 17 just adjusted to delivery only slightly impairs the air-charge supply to the internal-combustion engine but an influence is present. The arrangements described below and illustrated in Figures 3 and 4 are intended to obviate the impairment of the air-change supply to the internal-combustion engine. The difference with respect to the apparatus according to Figure 1 is a different source for the gas fed to the respective no-load delivery air-charge compressor. In the apparatus illustrated by way of example in Figure 3,

instead of compressed air, exhaust gas taken from the exhaust pipe 32 of the exhaust-driven turbine 15 is supplied. The gas-supply duct 25′ provided therefor leads from the exhaust pipe 32 to the distributor devices 27 and 28′. The exhaust-gas supply acts on whichever air-charge compressor 16 or 17 has just been adjusted to no-load delivery. This is the case for the air-charge compressor 16 in position "L" of the distributor device 27, corresponding to no-load and low load of the internal-combustion engine, and for the air-charge compressor 17 in position "T" of the distributor device 28′, corresponding to partial load of the internal-combustion engine.

The first air-charge compressor 16 under no-load delivery during no-load and low load of the internal-combustion engine has such a high no-load delivery rate that the entire amount of exhaust gas occurring during no-load and low load is drawn from the exhaust pipe 32, whereupon the pressure in the exhaust pipe 32 drops below atmospheric pressure. To prevent any atmospheric air from flowing back into the exhaust pipe 32, a non-return device 38 is provided in the exhaust pipe 32 downstream of the connection point for the gas-supply duct 25′. The non-return device 38 may be designed to be controllable automatically or by operating variables of the internal-combustion engine.

A gas cooler 40 disposed in the gas-supply duct 25′ has the effect that the volume of exhaust gas drawn in by the no-load delivery first air-charge compressor 16 is reduced by cooling. The pressure drop in the exhaust pipe 32 as a result of the drawing-in of exhaust gas is accompanied by an increase in the effective heat drop in the

the output of the exhaust-driven turbine 15 is then achieved.

In the apparatus according to Figure 4, the gas-supply duct 25" leads from the exhaust pipe 32 to the gas-supply duct 24′ upstream of the air-charge intercooler 39. The gas-supply duct 24′ is connected both to the feed connections 44, 45 to the distributor device 28" and to the feed connection 46 to the distributor device 27. With this line arrangement the air-charge intercooler 39 operates during no-load, low-load and partial-load conditions of the internal-combustion engine as a gas cooler for the exhaust gas drawn from the exhaust pipe 32. The additional exhaust-gas extraction compressor 41 is connected to the gas-supply duct 24′ upstrean of the air-charge intercooler 39.

In order to ensure, under full load of the internal-combustion engine, the transfer of air-charge from the first air-charge compressor 16 to the second air-charge compressor 17 by way of the gas-supply duct 24′ for two stage turbocharging, a controllable shut-off device 48 is disposed between the gas-supply duct 24′ and the exhaust-gas extraction compressor 41 and a non-return device 42 closing in the direction of the exhaust pipe 32 is provided in the gas-supply duct 25".

Instead of using the exhaust pipe 32 as the gas supply, it is also possible to tap the exhaust manifold 33 of the internal-combustion engine by way of the gas-supply duct 47 as shown in Figure 3. As a result of this measure the exhaust gas mass flow upstream of the exhaust-driven turbine 15 is slightly reduced, as is its effective output. Of course, this

exhaust-driven turbine 15, thereby increasing its effective output. The attainable power increase of the exahust-driven turbine 15 compensates for the power absorption, which still remains after application of all the other measures, of the first air-charge compressor 16 just changed to no-load delivery. In fact, the exhaust-gas cooling results in a lower power excess at the exhaust-driven turbine 15, the magnitude of which is dependent on the quantity of heat dissipated.

The pressure drop in the exhaust pipe 32 through a no-load delivery air-change compressor can be achieved only by the larger first air-charge compressor 16 which, during no-load and low-load conditions of the internal-combustion engine, is adjusted to no-load delivery. However, the no-load delivery rate of the smaller second air-charge compressor 17, which under partial-load conditions of the internal-combustion engine is adjusted to no-load delivery, is not sufficient to bring about a drop in pressure in the exhaust pipe 32 or excess power of the exhaust-driven turbine 15. An exhaust-gas extraction compressor 41 with drive means 43 is connected to the intake side to the gas-supply duct 25′ upstream of the gas cooler 40. The exhaust-gas extraction compressor 41 is operative whenever the second air-charge compressor 17 is adjusted to no-load delivery. The exhaust-gas extraction compressor 41 and the second air-charge compressor 17 then operate in parallel and draw exhaust gas from the exhaust pipe 32. Even under partial load of the internal-combustion engine a pressure drop is in this way obtained in the exhaust pipe 32. As described above for no-load and low-load operation of the internal-combustion engine, the desired increase in

does not produce any additional compensating effect for the air-charge compressor, which has just been adjusted to no-load delivery, in relation to the operating point "D" or "D'" in the compressor performance graph of Figure 2. However, because of the higher temperature and thus the lower density of the exhaust gas supplied at overpressure, the power absorption of the no-load delivery air-charge compressor 16 or 17 is reduced with respect to the operating points "A", "A'", "B", "B'", "C" and "C'" in the compressor performance graph according to Figure 2.

CLAIMS

1. A turbocharging assembly with controllable air-charge compressors (16, 17) for an internal-combustion engine, the air-charge compressors (16, 17) being in driving connection with an exhaust-driven turbine which cannot be cut out during operating periods of the internal-combustion engine, the operating condition (no-load delivery or delivery operation) of each air-charge compressor (16, 17) being determined by a change-over device (20) controlling the pressure connection, characterised in that upstream of the air inlet (13, 14) each controllable air-charge compressor (16, 17) has a multi-stage controllable distributor device (27, 28, 28′, 28′′), in that between the distributor device (27, 28, 28′, 28′′) and the air inlet (13, 14) there is provided a device (29, 30) for controlling the air flow acting on the rotor inlet, and in that at least one gas-supply duct (24, 25, 25′, 25", 47) is connected to the inlet side of each distributor device (27, 28, 28′,28′′).

2. A turbocharging assembly with controllable air-charge compressors for an internal-combustion engine according to Claim 1, characterised in that the gas-supply ducts (24 and d5) are connected to respective pressure connectors (18 and 19) of the air-charge compressors (16 and 17).

3. A turbocharging assembly with controllable air-charge compressors for an internal-combustion engine according to Claim 1, characterised in that the gas-supply duct (47) is adapted to be connected to the exhaust manifold (33) of the internal-combustion engine.

4. A turbocharging assembly with controllable air-charge compressors for an internal-combustion engine according to Claim 1, characterised in that the gas-supply duct (25′, 25″) is connected to the exhaust pipe (32) of the exhaust-driven turbine (15).

5. A turbocharging assembly with controllable air-charge compressors for an internal-combustion engine according to Claim 4, characterised in that a non-return device (38) is provided in the exhaust pipe (32) downstream of the connection point for the gas-supply duct (25′).

6. A turbocharging assembly with controllable air-charge compressors for an internal-combustion engine according to Claim 4, characterised in that a gas cooler (40) is disposed in the gas-supply duct (25′) between the exhaust pipe (32) and the discharge into the distributor device (27, 28′).

7. A turbocharging assembly with controllable air-charge compressors for an internal-combustion engine according to Claim 6, characterised in that an exhaust-gas extraction compressor (41) with drive means (43) is connected on the intake side to the gas-supply duct (25′) upstream of the gas cooler (40).

8. A turbocharging assembly with controllable air-charge compressors for an internal-combustion engine according to Claim 4, characterised in that the gas-supply duct (25″) is connected to the gas-supply duct (24′) downstream of

an air-charge intercooler (39).

9. A turbocharging assembly with controllable air-charge compressors for an internal-combustion engine according to Claim 8, characterised in that an exhaust-gas extraction compressor (41) with drive means (43) is connected on the intake side to the gas-supply duct (24′) upstream of the air-charge intercooler (39).

10. A turbocharging assembly with controllable air-charge compressors for an internal-combustion engine according to Claim 9, characterised in that a controllable shut-off device (48) is disposed between the exhaust-gas extraction compressor (41) and the gas-supply duct (24′).

11. A turbocharging assembly with controllable air-charge compressors for an internal-combustion engine according to Claim 8, characterised in that a non-return device (42) is provided in the gas-supply duct (25") downstream of the connection point to the gas-supply duct (24′).

12. A turbocharging assembly with controllable air-charge compressors for an internal-combustion engine according to Claim 8, characterised in that the gas-supply duct (24′) is connected simultaneously to the feed connections (44, 45) to the distributor device (28") and to the feed connection (46) to the distributor device (27).

13. A turbocharging assembly substantially as herein described with reference to and as shown in the accompanying drawings.

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